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Dynamic Correlation Technique and Model Updating on Go Kart Chassis Structure

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Abstract— **Chassis is a major component in a vehicle system. When the structure like chassis is excited at same value of natural frequency, resonance would occurred and damage the structure. The dynamic characteristic of go kart chassis such as the natural frequency and mode shape were presented using finite element analysis (FEA). The process of chassis design in the automotive was developed utilizing the advanced computer aided design. Structural modification and optimization are used to reduce component complexity, weight and subsequently the cost. Therefore, the model verification is needed before design decisions made in the FEA environment and it can be implemented in production with high confidence. Impact hammer test was carried out to validate the finite element (FE) model. Data analyzer was used to convert the response signal from the sensor which was in the time domain to frequency domain. The acquired result shows that the percentage of error for model updating is lower than initial FEA results. It indicates that the structural model updating technique is one of the method to give the better agreement between the experimental and computational results. Percentage of error reduced from 30% to below 11% for model updating of go kart structure. The purpose is to reduce the vibration as well as to improve the strength of the chassis.**

Keywords: Model updating, impact hammer test, finite element analysis, natural frequency and mode shape

I. INTRODUCTION

Early 50's, go-kart was first introduced to the public as a vehicle used not for transportation but for sports and recreation. Since go-kart was first invented over 40 years ago, analysis on the chassis structure has already begun and became more advanced until today. This parallel with high capability of computer technology, computer aided design and engineering tools. Structural analysis continues not just for safety and stabilization but to enhance the properties of the structure. Currently, the trend in chassis design involves the reduction of costs, weight of chassis and increase in safety efficiency are most important criteria. The consequence of a lighter chassis is a vehicle that has structural resonance within the range of typical rigid body vibrations of the go-kart subsystems. The vibration can be formed due to dynamic forces induced by the road irregularities, engine, passenger load and more. Thus under these various dynamic excitation, chassis will tend to vibrate and can lead to ride discomfort, ride safety and stability problems [1].

This paper focused on the dynamic correlation techniques which used to measure the accuracy of finite element representation of the go-kart chassis. Treating the chassis independently, analytical and experimental models were developed using finite element analysis and experimental modal analysis (EMA) techniques. The FE models are often correlated with experimental modal analysis (EMA) results in order to achieve high degree of confidence in the FE analysis. The EMA is a process where modal parameters such as natural frequency, mode shapes and damping ratio were extracted from the structures, experimentally. From 80's to 90's, the correlation and model updating were done based on the said modal parameters [2]. However, from late 90's onwards there is a shift of trend towards frequency response functions (FRF) based correlation and modal updating [3]. Model updating was performed to achieve a high degree of confidence in the FEA. In go-kart chassis development, the structural modification is one of the important stages. It is done through modifying the structure of chassis which result in reducing the vibration effect and improve the strength of chassis. The most common method used in structural modification is adding stiffener [4].

II. MATHEMATICAL MODELING

Finite element model can be obtained poles and frequencies from the finite element model, an eigensolution is performed on the mass and stiffness matrices. The equation of motion for a multiple degree of freedom system as in Equation (1):

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$$[m]\{\ddot{x}\}+[c]\{\dot{x}\}+[k]\{x\}=\{f(t)\} \tag{1}$$

where $[m]$ is mass matrix, $[c]$ is damping matrix, $[k]$ is stiffness matrix, $\{f(t)\}$ is forcing vector, $\{\ddot{x}\}$ is vector of acceleration, $\{\dot{x}\}$ is vector of velocity and $\{x\}$ is vector of displacement.

There are many tools are available for evaluate the correlation between FEA and EMA [5]. Modal assurance criteria (MAC) is most common method for evaluate the degree of correlation between any two vectors and The MAC can be expressed as in Equation (2):

$$MAC=\frac{\left|\phi_E^T\phi_A\right|^2}{\left(\phi_E^T\phi_E\right)\left(\phi_A^T\phi_A\right)} \tag{2}$$

where ϕ_E is experimental mode shape, and ϕ_A is analytical mode shape, ϕ_E^T and ϕ_A^T are the transpose of experimental and analytical mode shape respectively.

Model updating through the FE is to adjust the valued of selected parameters such that a reference correlation coefficient is minimized. Equation of model updating can be expressed as in Equation (3):

$$\{R_e\}=\{R_a\}+[S](\{P_u\}-\{P_o\}) \tag{3}$$

where $\{R_e\}$ is vector containing the reference system response, $\{R_a\}$ is vector containing the predicted system response, $\{P_u\}$ is vector containing the updated parameter values, $\{P_o\}$ is vector containing the given state parameter values and $[S]$ is sensitivity matrix.

III. FINITE ELEMENT ANALYSIS

Figure 1 shows the complete computer aided design for go-kart chassis which is developed utilizing the Solidworks software. The finite element model was developed using the tetrahedron element meshing technique. Based on previous findings, this node element was gave closer result to actual conditions [6]. The final chassis model consists of 20700 elements at 50% fine of meshing.

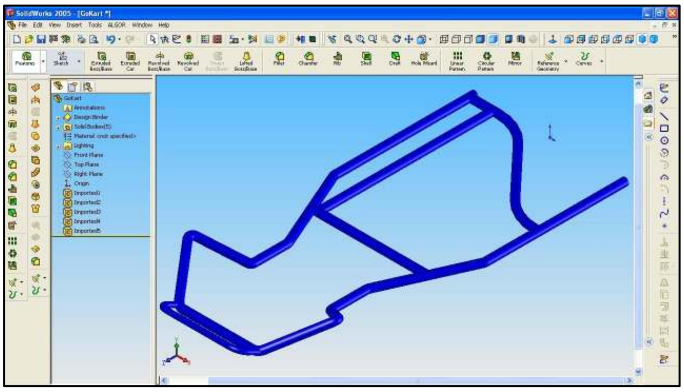
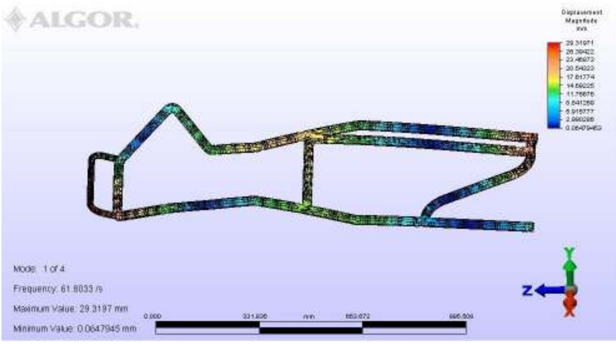
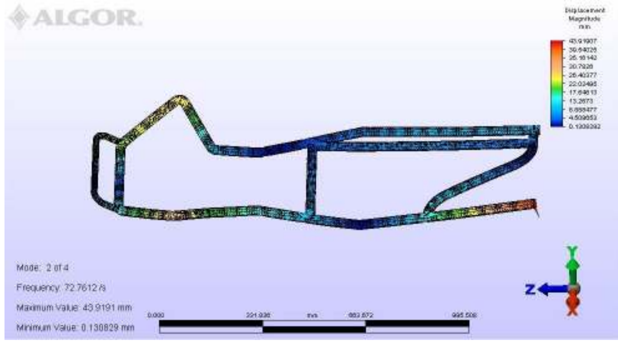


Figure 1: Model of Go-Kart Chassis

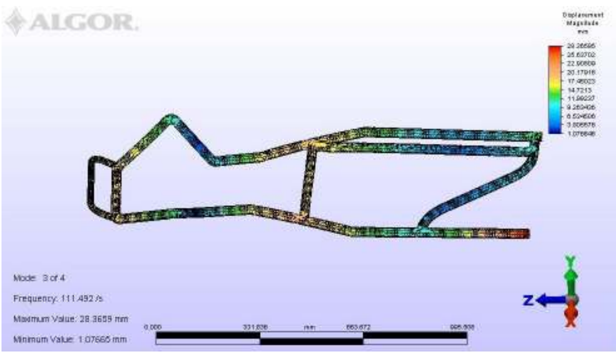
In prediction of analytical dynamic characteristics of go-kart chassis, the normal modes analysis was performed using commercial FEM Software, Algor. The free-free boundary condition was adopted in order to simulate the chassis's natural frequencies and mode shape vectors. Neither constraints nor loads were assigned to stimulate this free-free boundary condition. This analysis was perform on modal linear analysis with material as a steel (AISI 4130), lower cut-off frequency as 40 Hz and upper cut-off frequency as 200 Hz. The reason for setting frequency at 40 Hz is to avoid the solver from calculating rigid body motions which have the frequency at 0 Hz. Figure 2 shows the typical computational mode shape and natural frequencies of the go-kart chassis at 61.8 Hz at first mode, 72.76 Hz at second mode, 111.5 Hz at third mode and 125.5 Hz at fourth mode. The results show that the chassis experience a global vibration as the whole structure follow to vibrate.



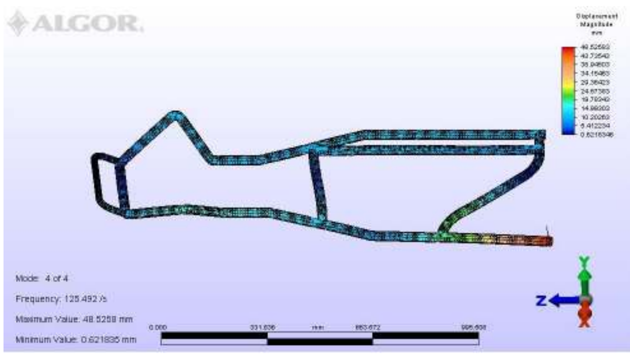
FEA first mode shape @ 61.8033 Hz



FEA second mode shape @ 72.7612 Hz



FEA third mode shape @ 111.492 Hz



FEA fourth mode shape @ 125.492 Hz

Figure 2: Natural Frequencies at 4 Mode Shapes

IV. EXPERIMENTAL MODAL ANALYSIS (EMA)

Experimental modal analysis is widely used in many engineering applications especially analysis involving vibration mechanism. Model parameters such as natural frequency, mode shape and damping ratio were extracted from the structure experimentally. Go-kart chassis was divided into 27 grid points where at this point, Frequency response functions were measured in the range of 0-400 Hz to identify modal parameters. These 27 points were chosen to give adequate spatial resolution to describe the global structural mode shapes.

Two excitation methods were implemented in the experimental modal test. The first method is known as impact hammer test, which was used to excite the 27 DOF on the chassis structure as shown in Figure 3. The force transducer was connected to the hammer at the bottom of the handle. The accelerometer or sensor was fixed at one reference DOF and the hammer was roving through all the 27 DOF. Data analyzer was used to convert the signal response which was in time domain to the frequency domain.

The second method was done with a fixed input location (in y-direction), with uniaxial accelerometers moved from point to point on the structure. This test known as shaker test and Figure 4 shows the set-up for shaker test. The boundary conditions were similar to FEM model where free-free boundary conditions were applied. There are some significant effects when using this method such as the locations of the accelerometer could affect the dynamics of structure [7].

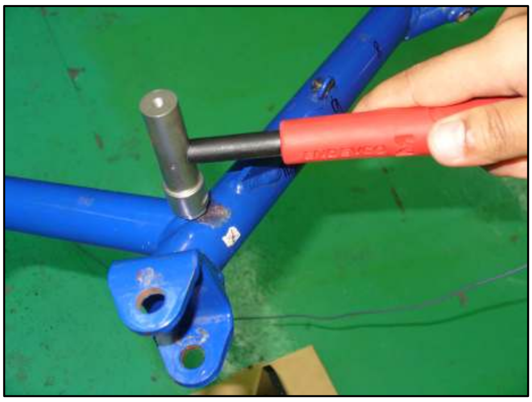


Figure 3: Impact Hammer Test

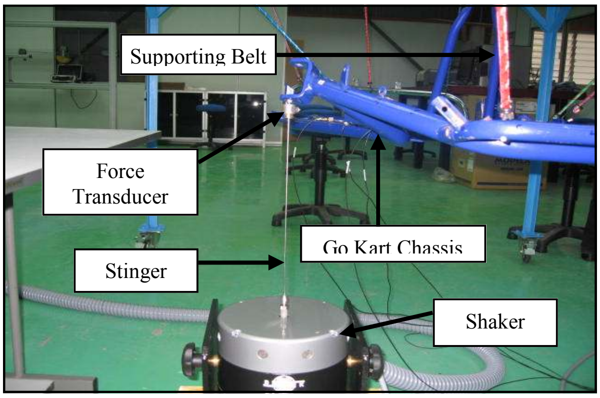


Figure 4: Shaker Test

Figure 5 and 6 show the superimposed FRF at 27 points for impact hammer and shaker test. Impact hammer test shows better FRF results because of mass loading are negligible. Figure 7 shows the experimental mode shape and natural frequencies of the go-kart chassis at 41.1 Hz at first mode, 61.8 Hz at second mode, 75.1 Hz at third mode and 83.2 Hz at fourth mode.

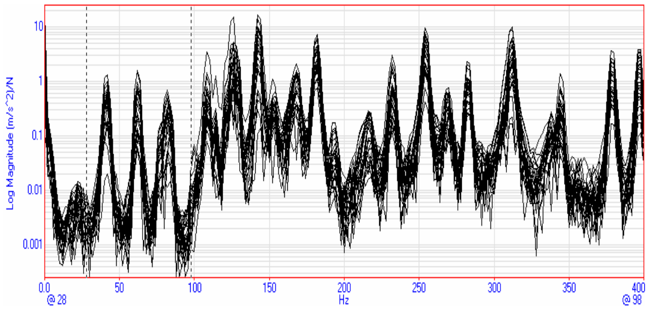


Figure 5: Superimposed FRFs by Impact Hammer Test

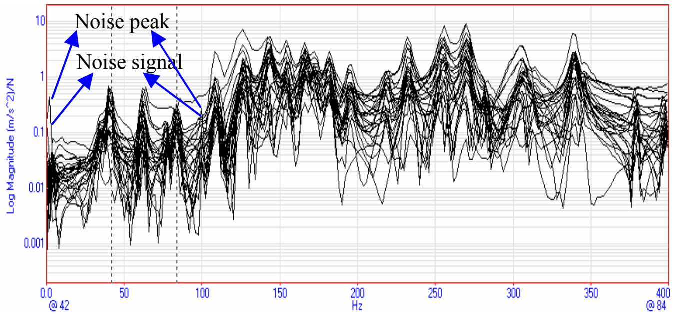


Figure 6: Superimposed FRFs by Shaker Test

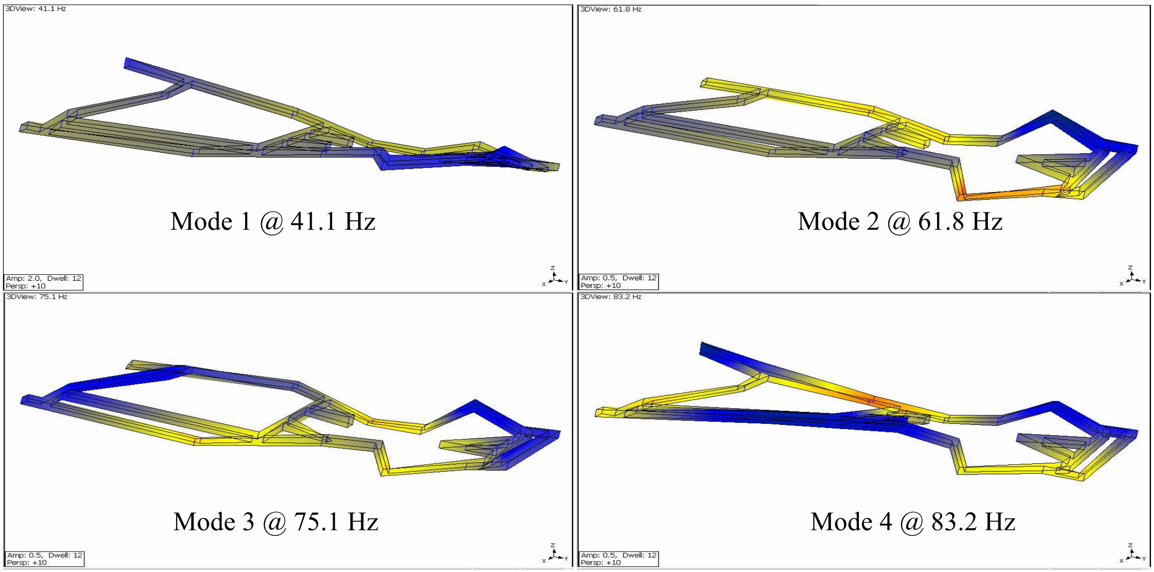


Figure 7: Experimental Mode Shapes and Natural Frequencies

V. CORRELATION BETWEEN FEA AND EMA

Table 1 is listed the natural frequency of different modes of the go-kart chassis extracted from both finite element model and experimental modal analyses including the Impact hammer and Shaker test. It can be seen from Table 1 that the Impact hammer and Shaker test results are closely related with each other. The natural frequency obtained from the test is not equivalent with each of FE mode frequency. It is noted that each FEA frequency is slightly higher than its matching test frequency, indicating that the stiffness of the FE model is greater than real structure.

Table 1: Comparison between EMA and FEA Natural Frequency results

Mode	Impact Hammer		Shaker		FEA Frequency (Hz)
	Natural Frequency (Hz)	Damping (%)	Natural Frequency (Hz)	Damping (%)	
1	41.1	0.019	41.4	1.20	61.8
2	61.8	0.014	62.6	0.503	72.7
3	75.1	0.101	77.8	0.816	111.5
4	83.2	0.0129	82.9	0.54	125.5

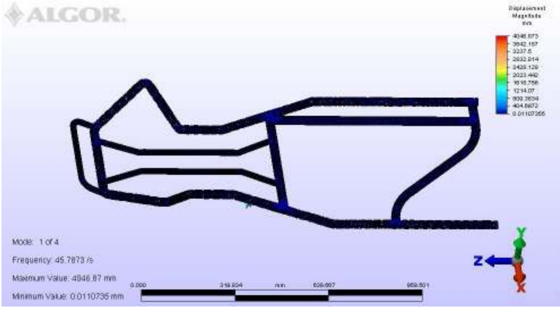
Correlation is processes to evaluate how close the frequency FE model with the experimental model. Discrepancies will always exist between the FE model and EMA model. This may attributed to the possibilities error in experimental data such as the measurements could have been carried out at an imperfect set-up noise and existence of inherent model parameter errors and model structure errors [8]. Modal Assurance Criteria matrix was performed and test would indicate how good the FE modes correlate with the test modes. The high MAC values (>75%) would show which FE mode shapes resembles to which test modes. Even though the error between EMA and FEA is quite high percentage but MAC results indicate that the mode frequencies were similar shapes.

Table 2: Mode Pairs with Frequency Difference

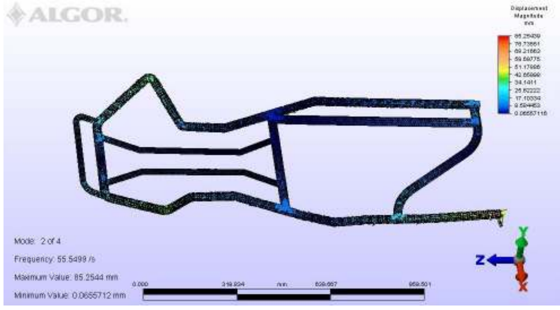
Mode	EMA Frequency (Hz)	FEA s Frequency (Hz)	Error (%)	MAC (%)
1	41.1	61.8	33.50	92.4
2	61.8	72.7	15.06	87.5
3	75.1	111.5	32.64	93.5
4	83.2	125.5	33.70	83.6

VI. Model Updating and Structural Modification

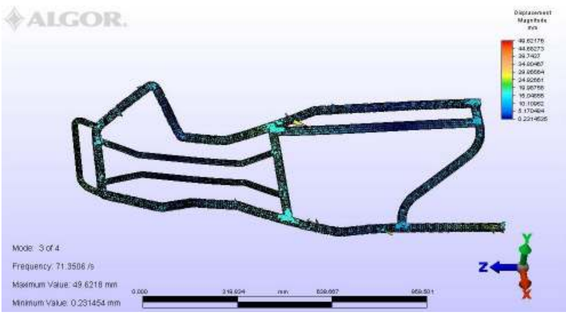
In order to justify the FE model into better agreement with the experimental data, the model updating and structural modification was needed. It is an important step in validation process that modify the values of parameters in FE model in order to create a reliable finite element model suitable for further analysis [4]. Some modification on the structure of go kart chassis was made and performs another analysis to determine frequencies and mode shapes. Figure 8 shows the finite element based modal analysis results i.e. the mode shape and natural frequency for structural modification of go-kart chassis.



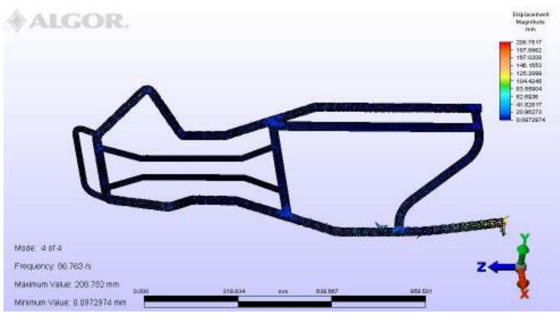
FEA first mode shape @ 45.7873 Hz



FEA second mode shape @ 55.5499 Hz



FEA third mode shape @ 71.3506 Hz



FEA fourth mode shape @ 86.7630 Hz

Figure 8: Structural Modification of Mode Shapes and Natural Frequencies

Table 8 shows the comparison between natural frequency initial and model updating FE results. The result shows that percentage of error for model updating is lower than initial FEA results. It indicates that the structural model updating technique is one of the method to give the better agreement between the experimental and computational results.

Table 3: Comparison between natural frequency initial and model updating FE results

Mode	EMA	Initial FE results		Model updating FE results	
	Frequency (Hz)	Frequency (Hz)	Error (%)	(Hz)	Error (%)
1	41.1	61.8033	33.50	45.7873	10.24
2	61.8	72.7612	15.06	55.5499	-11.25
3	75.1	111.492	32.64	71.3506	-5.25
4	83.2	125.492	33.70	86.7630	4.11

VII. CONCLUSIONS

In conclusion, the dynamic characteristics of go-kart chassis could be determined using FE analysis when the right element and method are used. However, due to model complexity, some simplification on design was done and large error could be expected. Therefore, for the model to be useful, EMA should be carried out to verify the real structure. For dynamic analysis, the FE model proved to have positive or strong correlation with EMA in the frequency and mode shape. Structural modification was carried out to get better result in term of mode shape and frequency. Percentage of error reduced from 30% to below 11% after some modification was doing on go kart structure.

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